

THRUST BEARING ANALYSIS, OPTIMIZATION AND CASE HISTORIES

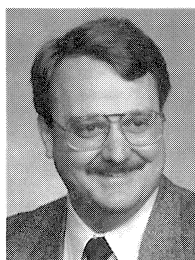
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ABSTRACT

Babbitted oil film thrust bearings as used in industrial turbomachinery are discussed. The majority of the presentation covers the higher performance tilting pad thrust bearings as found in most critical rotating equipment. Topics covered include types of thrust bearings, terminology, analysis, and application of thrust bearings. A discussion of common design options available is presented, followed by a couple of case histories, used to illustrate the analysis procedure and subsequent effects of the optimized redesigns.

INTRODUCTION

Most critical rotating equipment in the refining, chemical, and petrochemical industries contain thrust bearings to counteract internal axial loads generated by pressure differentials in the machine. Coverage is limited to babbitted oil film thrust bearings as they are most commonly found in critical turbines and compressors. Thrust bearing types, terminology, and operating limits are discussed. An analysis tool called THRUST [1, 2] is then introduced. Application of the program is used to illustrate various design options available with tilting pad thrust bearings to increase load capacities. Detailed coverage of two case histories is then presented to illustrate the application of the program to actual problem bearings.

THRUST BEARING TYPES

The drawing in Figure 1 shows a flat land thrust plate used for low thrust load applications. The babbitted thrust face has radial oil

distribution grooves. The babbitted face is flat and can only support loads below 75 psi [3] (thrust bearing loading is calculated by dividing the total axial applied load by the babbitted thrust face active area.) This style bearing is found most often in "bumper" thrust type applications where sustained axial loads are not anticipated (such as with horizontal electric motors or inactive thrust bearings in some smaller turbines and compressors.)

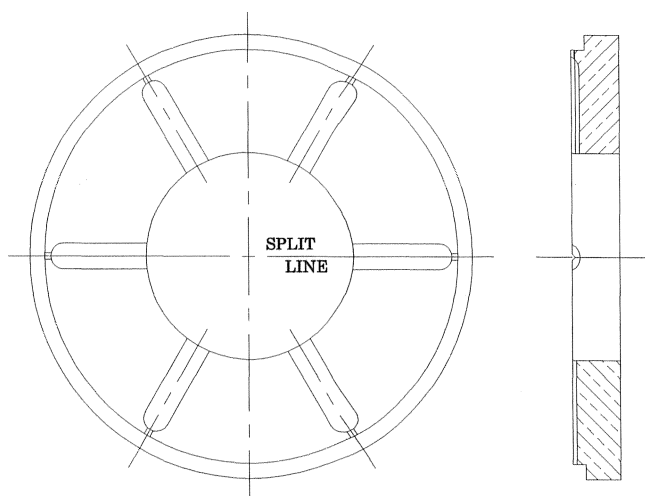


Figure 1. Flat Land Thrust Plate.

The next step up is the taper land thrust bearing (Figure 2). This bearing's use is limited to about 200 psi maximum [3] and is best applied to constant load/speed applications where the babbitt face can be optimized for a single application. The taper portion is optimized for this single load case while the land portion (about 20 percent of total bearing area) is dedicated to be used as a bumper thrust bearing (a surface to bump against when setting float or rotor position for example, or for low speed operation). There are several variations on this style bearing from straight tapers (taper from leading edge to trailing edge) to "compound" tapers (tapering from inside diameter (ID) to outside diameter (OD) and from leading edge to trailing edge). These bearings are direction sensitive and cannot support any appreciable loads in reverse rotation.

The tilting pad thrust bearing (Figure 3) can support loads over a range of speed and load conditions. The pad can tilt as required to support the load applied. Basically the tilt pad thrust bearing is similar to the compound taper thrust bearing except the pad tilting optimizes the babbitt face for any combination of speed and load, instead of just one specific case, as with the fixed geometry taper land bearing. Also, the flat land area is not required since the pad will orient itself to a flat position when required. The bearing as shown in Figure 3 is referred to as a "non equalized tilting pad thrust bearing" to differentiate it from the "self equalized" bearing as shown in Figure 4.

addressed in most thrust bearing upgrades is reducing the maximum bearing temperature. Oil film thickness, film pressure, and mechanical factors also need to be addressed, but they are usually well within required design constraints (especially if the pad temperature is within acceptable levels).

One of the easiest ways to cool down a hot running bearing is by upgrading the pad backing material from steel to a copper alloy. The high heat conduction properties of the copper pulls heat from the babbitt face where it can do the most damage. Several papers have been published on the impact of pad material on thrust bearing performance. One such paper [4] includes test results that can be correlated with THRUST. A "catalog" six inch thrust bearing was tested with steel pads and copper alloy pads. The same conditions were analyzed and the results plotted in Figure 6.

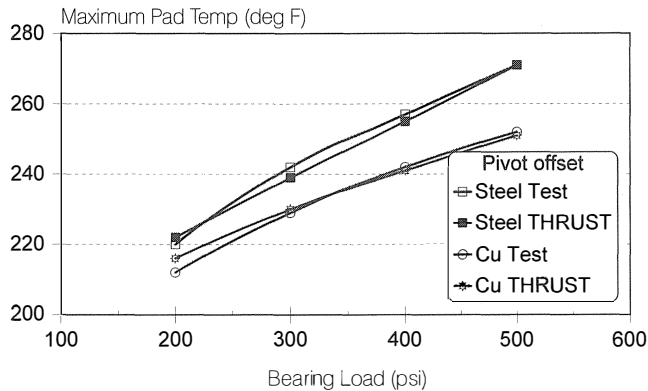


Figure 6. Effect of Pad Backing Material on Babbitt Temperature, Comparison of Test Data to THRUST.

The top two lines in Figure 6 are the test results and analysis results for the bearing with steel pads. As shown, the correlation between test and analytical data are very good. The bottom two lines are the same conditions with the copper alloy pad material, again note the good correlation.

Inspection of Figure 6 illustrates the temperature reduction possible with the pad backing material upgrade. Note that as the loading increases the temperature spread between the steel and copper alloy pads also increases. This implies that the pad material upgrade yields even better results at the higher temperatures. Indeed, for this case, at 500 psi loading a 20°F drop in temperature is attained.

Effect of Offset Pivots

Another dramatic design variable that can be optimized is the circumferential pivot location. Most "catalog" thrust bearings are supplied with pivots at a 50 percent offset (centered circumferentially). Moving the pivot towards the trailing edge forces the leading edge film thickness to be greater, thereby bringing more cooling oil onto the thrust face, resulting in lower operating temperatures.

The plot in Figure 7 is used to illustrate the correlation of the program with test data [4] along with demonstrating the impact of the offset pivot configuration on performance. The top two lines are center pivot (50 percent offset) results for the test and from analysis. Note the very good correlation between the test data and the analysis. The bottom two lines are results for the 60 percent offset pivot bearing. Here the correlation is off a little, with the analysis predicting higher temperatures (on the conservative side). This is attributed to the fact that the temperature sensor in the pad may not be at the hot spot in the bearing when the offset pivot is utilized. It has been found that moving the temperature sensor

downstream circumferentially, with offset pivot bearings, may be in order to more accurately capture the hot spot in the pad.

For consistency, most thrust bearings are equipped with bearing temperature sensors at the "75-75" location [5]. As per API 670 third edition, Section 4.1.5.2.2, the 75-75 location is defined as follows:

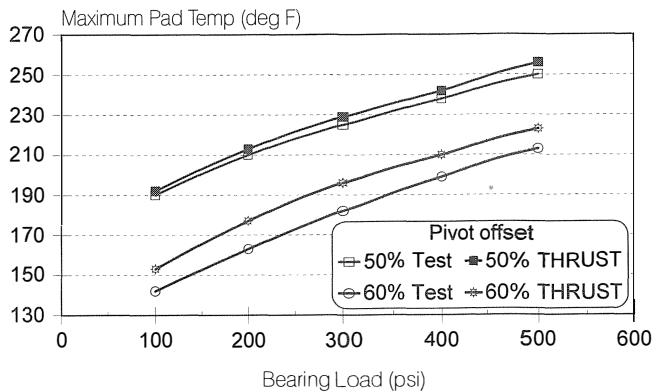


Figure 7. Effect of Pivot Offset on Babbitt Temperature, Comparison of Test Data to THRUST.

"Thrust bearing temperature sensors shall be placed at 75 percent of the pad width radially out from the inside bearing bore and at 75 percent of the pad length from the leading edge."

The sketch in Figure 8 illustrates thermocouple (TC) or RTD placement in tilting pad thrust bearings. Note that the sensors should be imbedded in the pad backing material, not the babbitt. Also note that the sensor should be 0.060 in to 0.100 in from the babbitt face but no closer than 0.030 in from the bond line. As an example, for babbitt 0.060 in thick, the sensor should be 0.090 to 0.100 from the babbitt face (0.060 babbitt plus 0.030 minimum backing material).

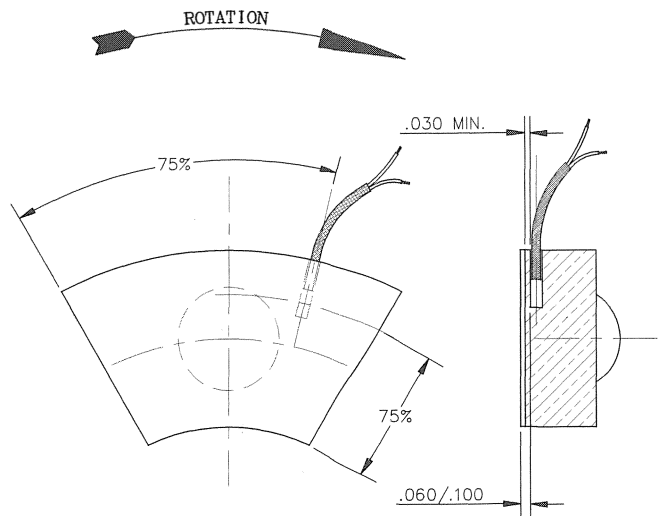


Figure 8. Temperature Sensor Location in Thrust Bearings as Per API 670.

Effect of Directed Lubrication

The means of lubricating the bearing does not have as pronounced effect on bearing metal temperature as does pad material and pivot offset. However, with hot running bearings, a 10°F or so drop in bearing operating temperature can be achieved, by upgrading from a standard flooded lube configuration to a directed lube design.

In the conventional flooded lube design oil is introduced at the bore of the bearing, travels up the face of the thrust collar, and is discharged through a drain port at the top of the bearing chamber, at the collar OD. In this case, the entire chamber is flooded with oil and churning of the oil by the collar introduces heat to the bearing beyond the shearing of the oil film. In fact, the amount of heat generated by churning is often on the same order of magnitude as the heat generated by the shearing of the oil film. As a result of this, power savings can be realized by upgrading to directed lube bearings. Indeed with large utility sized bearings (25 in pad OD and above) power savings of several hundred horsepower can be realized.

A directed lube bearing minimizes this churning by introducing the oil through a series of orificed nozzle blocks located between pads. These blocks supply the cool inlet oil directly to the leading edge of each pad thereby eliminating the need to flood the chamber with oil. In this case the chamber has generous drain provisions at the bottom, located at the collar OD.

The effect of directed lubrication was analyzed with very good results. New [6] summarizes the results of a series of tests run with a 4-7/8 in bearing, comparing flooded to directed lubrication. The plots in Figures 9, 10, 11, and 12 summarize the results of this testing along with correlation to THRUST. The plot in Figure 9 is the test data showing a 10°F drop in metal temperature for loads from 150 to 550 psi. The same plot is shown in Figure 10, except with the analysis results plotted. Note that the analysis also predicts the 10°F drop. The plots in Figures 11 and 12 compare the analysis with the test data for the flooded and directed lube cases, respectively. As shown the analysis slightly overpredicts the metal temperature. This may be attributed to the fact that the tests were run with ISO 68 oil and correlation of the program with this heavy an oil was not attempted. The trending was the same thereby increasing confidence in the program.

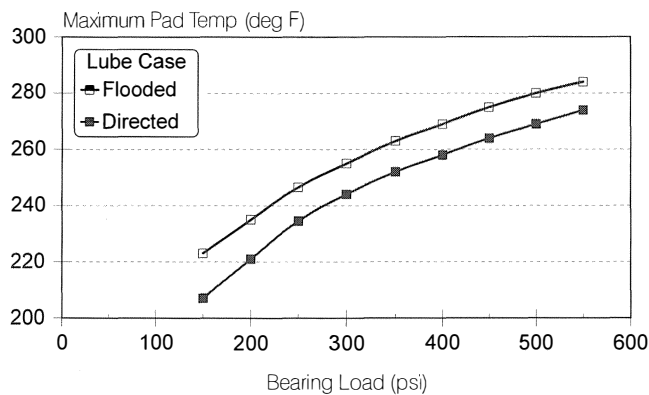


Figure 9. Directed Lube Vs Flooded Operation Effect on Babbitt Temperature, Published Test Results.

These cases of three different thrust bearing configurations illustrate the accuracy of the program. Conveniently, they also help explain the impact of three design variables on thrust bearing performance. These parameters were concentrated on because they are three improvements that can be made relatively easily as part of an upgrade package for a given application.

CASE HISTORIES

Two case histories are now presented that illustrate the use of the program and some of the design concepts presented.

Repeated Failure of Nonequalized Tilting Pad Thrust Bearing in a Barrel Compressor

This case is pulled from the Ammonia industry, where a class of centrifugal compressors used in syn gas service have experienced

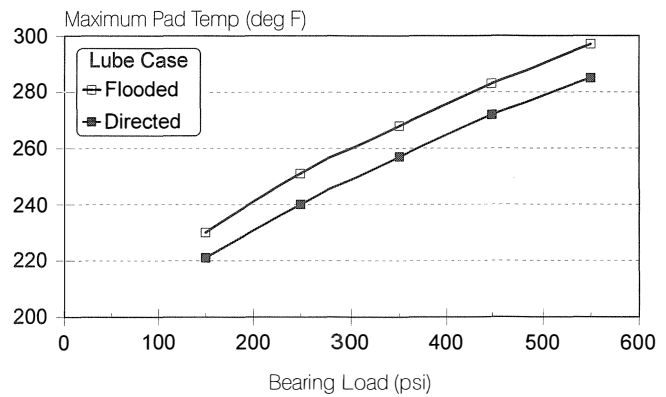


Figure 10. Directed Lube Vs Flooded Operation Effect on Babbitt Temperature, THRUST Analysis.

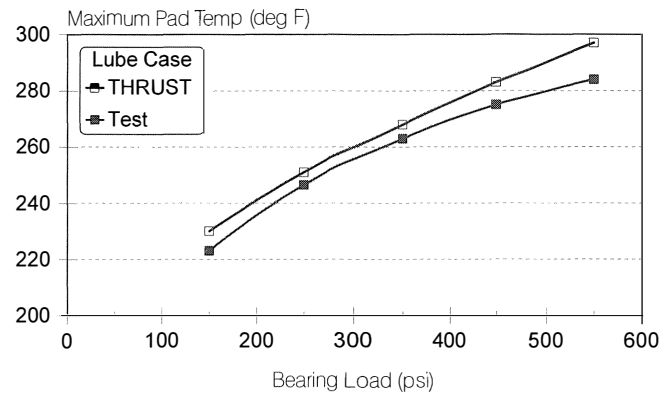


Figure 11. Flooded Lube Babbitt Temperature, Comparison of Test Data to THRUST.

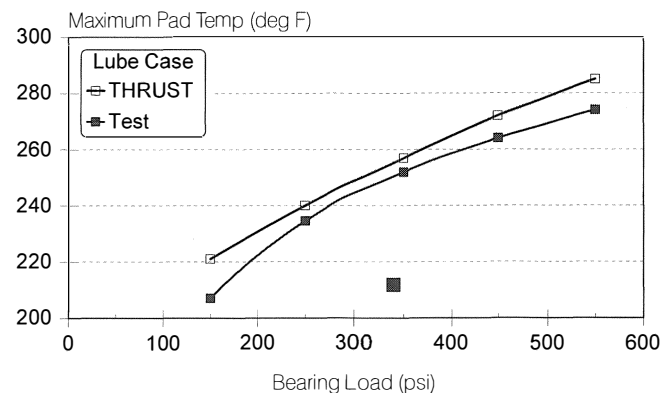


Figure 12. Directed Lube Babbitt Temperature, Comparison of Test Data to THRUST.

several thrust bearing failures. The problem is with the active bearing, a non equalized tilting pad thrust bearing. RTDs, that were installed several years ago, basically confirm that as these plants have been pushed harder, so have the thrust bearings in these machines. RTD readings of 235 to 240°F were common. In this case, it was observed, through failure analysis and RTD readings, that the thrust load was evenly distributed over the face of the bearing, implying that an upgrade to an equalized bearing was not necessary.

THRUST was used by analytically increasing the load on the bearing until the steady state observed operating temperature was achieved. This resulted in a calculated load of 10,250 lb (464 psi)

on the thrust bearing. The design option investigated involved changing the pad material from steel to chrome copper (copper used for its heat conduction properties with the chrome added for stiffness and strength.) Analysis with the new pad material properties indicated that a 20 to 25°F drop in metal temperature could be realized with this upgrade.

A plot from this analysis is shown in Figure 13. Plotted is the maximum predicted pad temperature vs bearing loading for both the steel and copper alloy pad backing materials. Note that with steel pads at 240°F the calculated load is 464 psi. Rerunning with the copper alloy pad material shows a drop to 217°F predicted. Note also that the spread between the two curves increases with increasing load, as shown earlier.

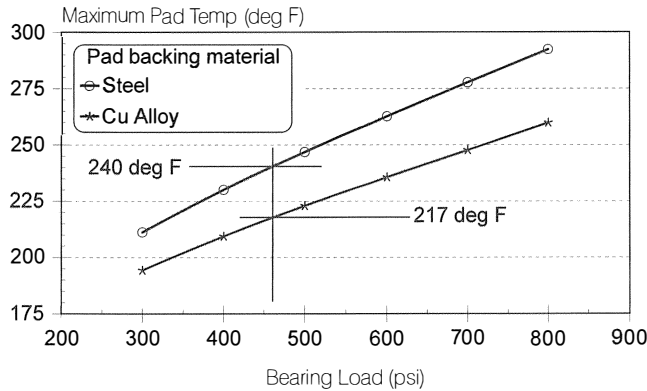


Figure 13. Case History #1, Predicted Effect of Pad Backing Material on Babbitt Temperature.

With the steel pad design the pivot was machined into the back side of the pad. This design was unacceptable with the relatively soft copper alloy pad so a pivot redesign was required. Since it was desired to only change the pad material and not the fundamental pivot design, a steel rib was attached to the pad backside for the pivot. This rib was hardened and ground with the proper radius for the pad pivot. Figure 14 is a drawing showing this basic design upgrade. Figures 15 and 16 are photographs further illustrating the design concept by comparing the front and back side of a failed steel pad with the upgraded copper alloy pad.

The pads were retrofitted (no work to the pad retainer was required) and installed during the summer of 1991. No other factors that would influence the pad loading were altered. As predicted thrust bearing temperatures dropped 20 to 25°F on all upgraded bearings.

This clearly demonstrates one of the major design variables available to combat hot running thrust bearings. To date, this upgrade has been successfully applied over a dozen times in three different ammonia plants with similar results.

Failure of an Equalized Thrust Bearing [7]

Original Failure

Figure 17 is a photo of a failed thrust bearing. The bearing was an equalized directed lube design with copper alloy thrust shoes. The pads wiped so hard that copper actually metallurgically bonded to the thrust collar. This bearing only ran three weeks. Operating metal temperatures were about 215 to 220°F and all four instrumented pads read within 7° of each other (indicating load equalization).

Figure 18 is a close up of a single pad. The hole in the center is where the pads were drilled so the pad support could be "robbed" to shorten the lead time to make another set of pads. Note that

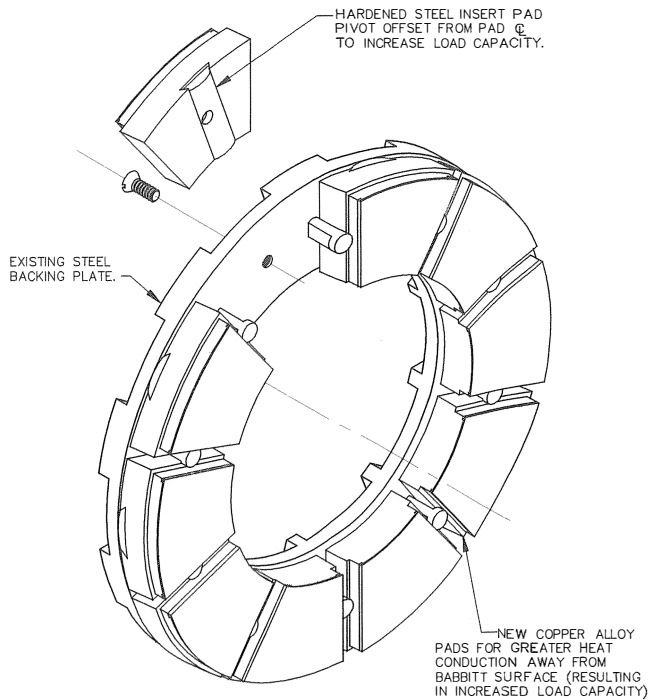


Figure 14. Case History #1, Upgraded Bearing Design Features.

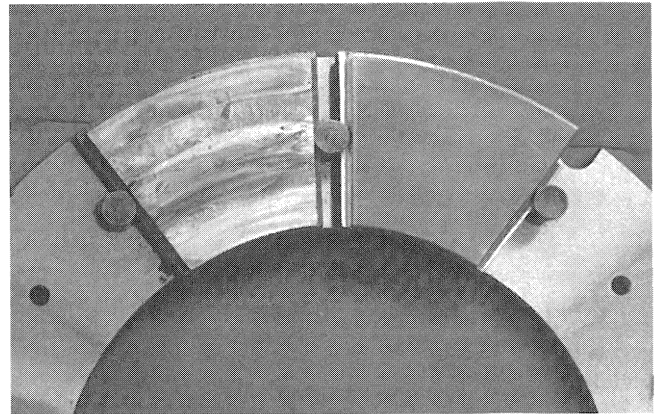


Figure 15. Case History #1, Photograph Comparison of Failed Pad Design to Upgrade Pad Design, Looking at Babbitt Face.

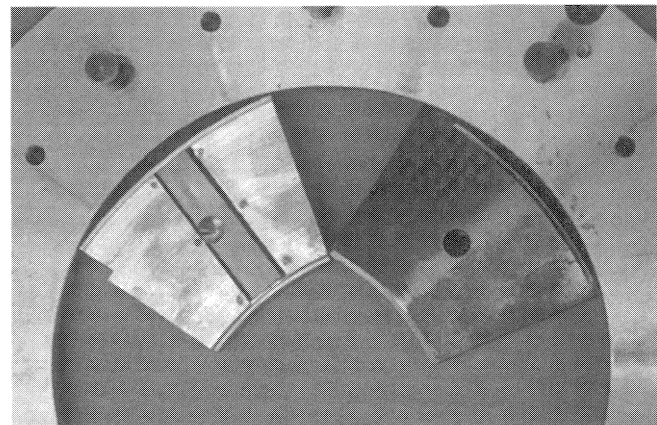


Figure 16. Case History #1, Photograph Comparison of Failed Pad Design to Upgrade Pad Design, Looking at Pivot Designs.

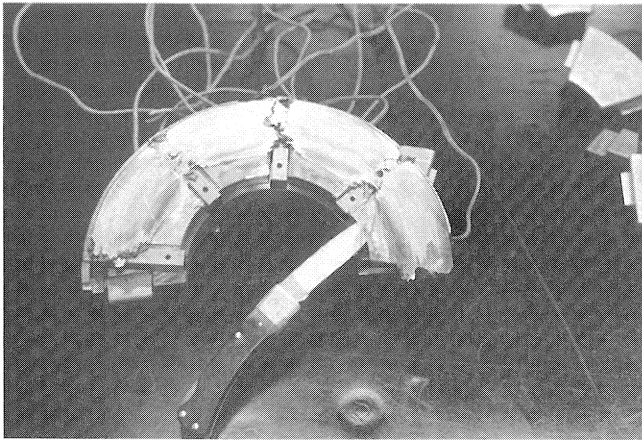


Figure 17. Case History #2, Photograph of Catastrophic Bearing Failure.

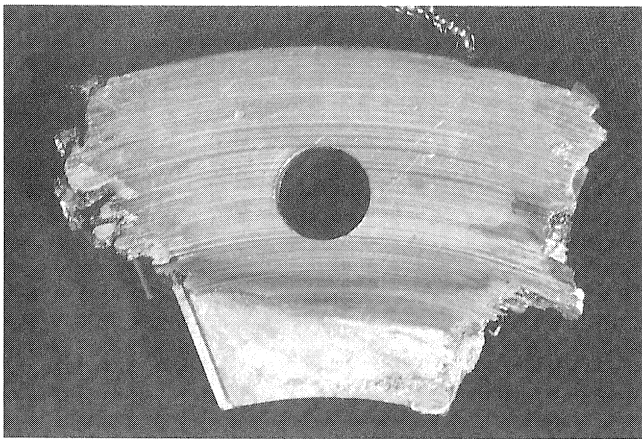


Figure 18. Case History #2, Close Up Photograph of a Failed Pad from Figure 17.

below the copper smear the pad is still “tinned” indicating a good babbitt bond. Also note that the only babbitt on the pad is that which was jammed into the dovetail on the leading edge of the pad.

Second Failure

A shot of the next bearing to come out of the machine is shown in Figure 19. This bearing only ran for about two weeks. Notice that the trailing edge of each pad shows signs of damage.

Successive close ups of the bearing are shown in Figures 20 and 21. Note in Figure 20 that the thermocouple (TC) is on the *leading edge* of the pad. Reinspection of Figure 18 (the previous failure) further demonstrates the improper TC installation. The babbitt redeposit in the dovetail is on the leading edge which is also the same side the TC exits. Obviously, the hot spot is on the trailing edge. This explains the relatively low metal temperature readings (215 to 220°F). The damage as shown in Figures 19, 20, and 21, and the leading edge temperature, indicate that the trailing edge was probably seeing temperatures of about 265 to 270°F. This was verified by running THRUST with increasing load until the point at the actual TC location predicted 215 to 220°F. Output from the program (Figure 22) shows that 215 to 220°F at the leading edge TC location equates to 265 to 270°F at the hottest part of the pad.

Optimized Bearing

Figure 23 is a photo of a redesigned bearing to help solve the failure problem. This bearing has 14 percent more surface area,

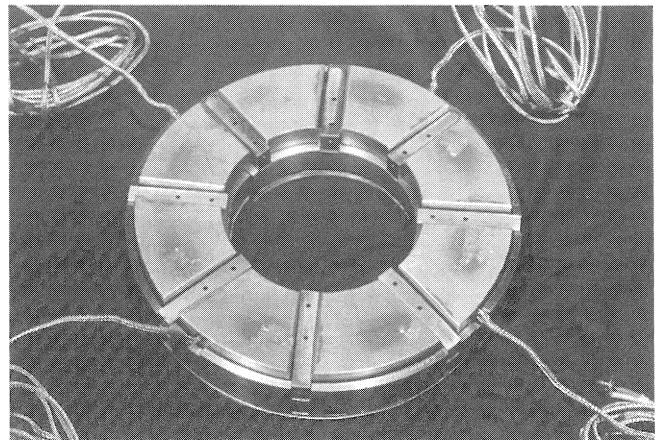


Figure 19. Case History #2, Photograph of Bearing Near Full Failure.

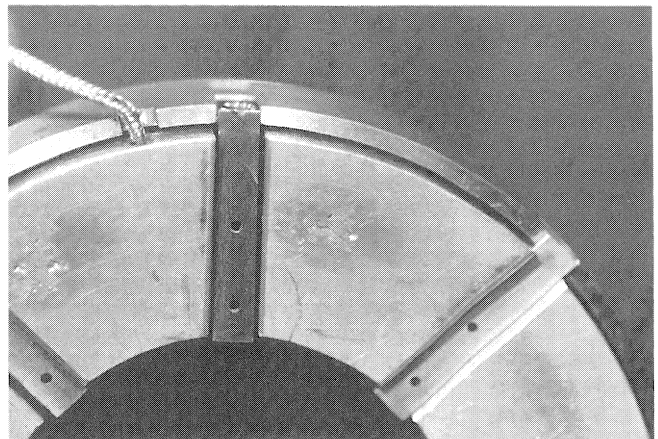


Figure 20. Case History #2, Close Up of Some Pads from Figure 19, Note TC Location.



Figure 21. Case History #2, Close Up of Damaged Corner of One Pad from Figures 19 and 20, Note Severe Babbitt Degradation.

more efficient oil inlet nozzling, and an offset pivot design.

The increased area acts to lower the psi loading on the bearing. The area increase was obtained by decreasing the bearing bore by 0.5 in, and by using thinner nozzle blocks thereby affording a larger arc length pad. Unfortunately increasing the pad outside

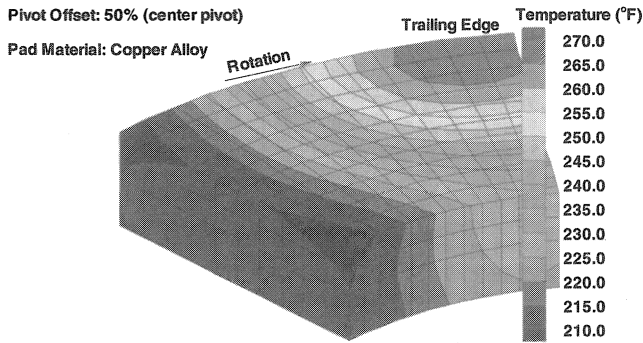


Figure 22. Case History #2, Output from THRUST for Original Bearing Design, Note That at the Actual Installed TC Location the Predicted Temperature is 215 to 220°F, While at the Hottest Part of the Pad the Predicted Temperature is 265 to 270°F.

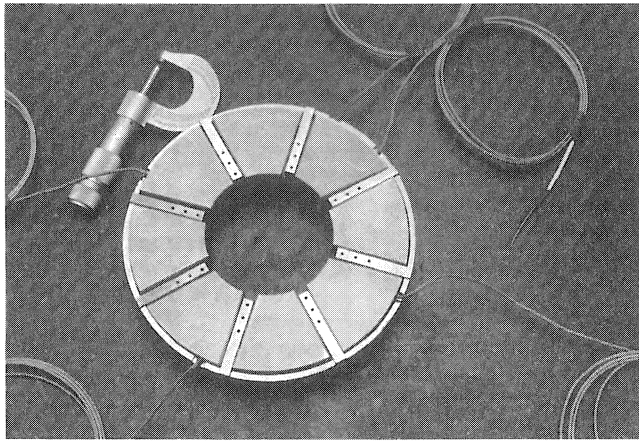


Figure 23. Case History #2, Photograph of Fully Optimized Upgraded Bearing.

diameter (OD) was not an option, this would have yielded a greater area increase and a higher mean velocity, which both would have acted to increase load capacity even further.

The renozzling did not increase the oil flow but added an orifice and reduced the orifice sizes to 1/16 in. This acted to give the oil greater velocity and, therefore, a better scouring action removing the hot oil carry over from the previous pad. It was found earlier that increasing flow by increasing oil pressure did not lower metal temperatures, therefore it was decided that additional flow was not required. This implies that the bearing was operating on the asymptotic portion of the flow vs temperature plot.

The major design upgrade was with the offset pivot configuration. Pivot offset is measured from the pad leading edge. A center pivot bearing has an offset of 50 percent. Most offset pivot bearings have offsets of about 55 percent. With this bearing a 65 percent offset was used.

When evaluating thrust bearing performance the parameters that are often considered are maximum metal temperature and minimum oil film thickness. Other parameters to consider are maximum oil film pressure and component stress levels.

With the bearing under consideration the stresses are all within acceptable levels. The redesign therefore needed to minimize metal temperature, maximize oil film thickness, and minimize the maximum oil film pressure. Calculations revealed that the metal temperature minimum occurred at an offset of 75 percent, minimum film thickness maximum occurs at 70 percent, and maximum film pressure minimum occurs at 65 percent.

The 65 percent offset was chosen considering the above and the physical design limitations. Figure 24 is a shot of the back side of a pad showing the pivot offset. Overall, a temperature reduction of 80°F was anticipated due to the various design changes.

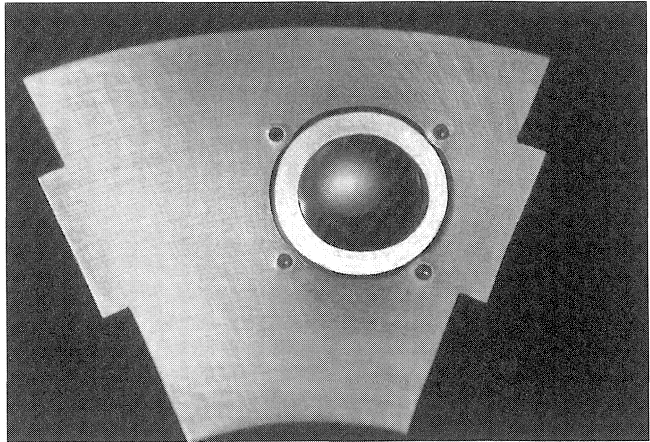


Figure 24. Case History #2, Photograph of Backside of One of the Upgrade Pads, Note the Extreme Pivot Offset.

Also note in Figure 23 that four pads are again instrumented but now on the trailing edge. One of the four pads is also instrumented on the leading edge, so that a comparison could be made with the metal temperatures from the previous two bearings. Output from the analysis for the optimized bearing is shown in Figure 25. The load and speed were kept the same as the run output in Figure 22. The increased area lowered the unit loading from 700 psi to 612 psi. As shown in Figure 25, a substantial reduction in temperature is predicted. Indeed, at the leading edge TC a temperature of 145°F is predicted. Indeed, at the 75-75 location a temperature of 175°F is predicted.

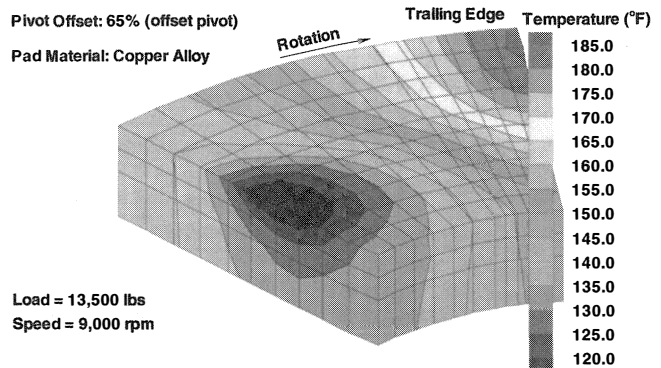


Figure 25. Case History #2, Output from THRUST for Upgraded Bearing Design, Note That at the Correctly Installed TC Location the Predicted Temperature is 175 to 220°F, While at the Leading Edge (Where the TC Was Originally Installed) the Predicted Temperature is 145°F.

Resulting Performance

The machine came up and ran for nine months before being removed from service for replacement by a new turbine. Metal temperatures (on the trailing edge!) read below 180°F, very close to predicted. The leading edge TC registered a reading of 135°F, 10°F lower than predicted. Overall very good correlation was found between the program and this actual case history.

When the optimized bearing was installed no other changes to the machine were made. The turbine had been down and back up three to four times after the initial installation to recharge the catalyst. All TCs read below 200°F during the entire remaining life of the machine, including the transients associated with bringing the process up and down.

CONCLUSION

The intent herein was to demonstrate a state of the art thrust bearing program and discuss thrust bearing design options available to solve installed thrust bearing problems. Three major design variables were discussed and evaluated with the program. A couple of case histories brought all of this together in a real world setting further demonstrating the effectiveness of these design upgrades and the accuracy of the program.

ACKNOWLEDGMENTS

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The author would also like to acknowledge the tremendous assistance of Ted Brockett of the University of Virginia, the author of THRUST. His patience with the author was very appreciated.

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